

The Free Piston Stirling Principle as Prime Mover for Alternant Hydraulic Propulsion Systems

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Abstract The objective of the presented research work is the demonstration of the working principle as energy efficient and environmental friendly, low cost vehicle propulsion system of the free piston Stirling principle coupled with the alternant hydraulic transmission in order to use the alternant displacement of the Stirling principle, also to the hydraulic transmission and convert the alternant movement of the liquid using a free wheel based hydraulic motor. The design methodology considers the advantages of efficiency, noise and low emissions of Stirling principle, that can be further improved by a free piston solution and the advantages of a direct alternant energy transfer to the liquid column that is converted in a free wheel based hydraulic motor. An analytical and 1D multi-domain model is used in order to demonstrate the running principle, performance and parameter influences that allow the simulation of the combined Stirling principle with alternant hydraulic transmission integrated on a vehicle. For the heat transfer and flow simulation of the Stirling prime mover, 3D CFD software is used to simulated the temperature distribution. Experimentally the solution was demonstrated on a test rig and integrated on a ATV type demonstration vehicle equipped with the alternant hydraulic propulsion system. Flow, thermal and mechanical parameters

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are measured to evaluate overall system performance. The former research work of the authors was focused on the demonstration and performance evaluation of the alternant hydraulic propulsion, being demonstrated its efficiency (up to 95 %), adaptability and low mass qualities. Main advantages of the Stirling principle may be improved by using a proper transmission. Coupling also the transmission based on the same alternant principle is a low cost, simple, efficient way to maintain the advantages of the Stirling energy conversion in the same time with an application of the hydraulic transmission that works also alternant in a more efficient way.

Keywords Vehicle • Propulsion • Stirling • Hydraulics • Simulation

1 The Technical Solution

The objective of the presented research work is the demonstration of the working principle as energy efficient and environmental friendly, low cost vehicle propulsion system of the free piston Stirling principle coupled with the alternant hydraulic transmission. The basic concept is, that primary energy converter and transmission have to be fitted and accorded one to each other, and functions have to be transferred from a system to the other in order to assure overall system efficiency.

The general concept is based on using the Stirling principle as primary energy source, able to convert the thermal energy resulting from fuel combustion. Fuel is still a very important high energy density carrier, at least for the foreseeable future, when significant measures for CO₂ reduction are stringent necessary.

The Stirling engine, considering the synthesis presented in [1], pp. 159–160, has as major advantages: continuous, external combustion is associated with low emissions and noise, free choice of the fuel, no combustion of lubrication oil, more simple injection system, potential for lower consumption due to improved cycle efficiency. The theoretical efficiency is affected by the continuous movement of the pistons in opposite to the discontinuous movement of the ideal cycle. Other factors that affect real to ideal cycle differences are isothermal processes, limited heat transfer in the regenerator, internal heat transfer. The disadvantages of the conventional Stirling engine are the costly heavy construction, where a significant contribution has the mechanical mechanism of fluid transfer and for gathering the rotational movement. Difficult control for load adjustment, large cooler areas, running up (30 s) and shut off (120 s) behaviour are factors considered also as disadvantages that limit the vehicle applications being necessary supplementary research, considering that proven applications in combined heat and power plants are working at 94 % efficiencies, with low asset and maintaining costs.

Analysing the disadvantages, it was considered that an improvement of the Stirling design would be the avoidance of the mechanical mechanism and direct energy transfer to the fluid power based transmission, the transfer of the load control, start up and switch off functions also to the fluid power transmission by

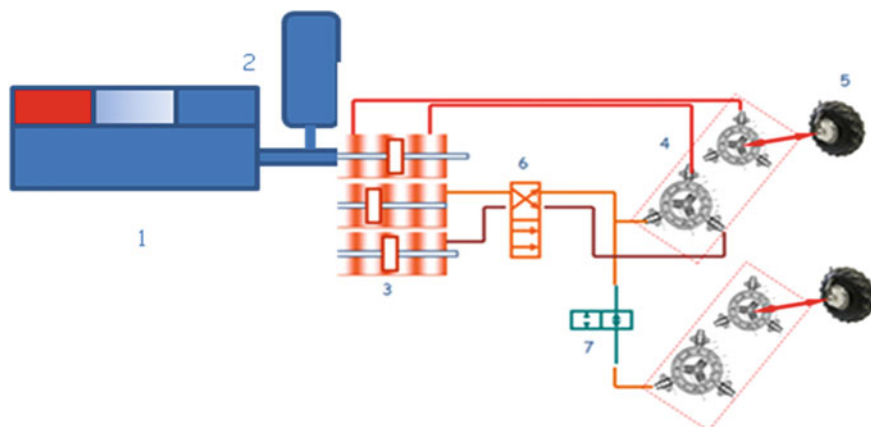


Fig. 1 Working principle of the Stirling alternant hydraulic propulsion system

using an accumulator that would cumulate several functions: assist start up and switch off, assure control functions of power output, allowing the engine to run at constant parameter, allow a free control of piston movement, including constant volume run, allow brake energy recovery.

At transmission level the fluid power based option was chosen due to its advantages for power density, precise control, and simple layout for the propulsion when multiple hydraulic actuators have to be fed to obtain the propulsion functions of integral wheel actuation, considering running during cornering, synchronisation of wheel speed when excessive slip occurs. The typical disadvantage of lower efficiency has been considered to be improved by using alternant hydraulic solutions based on a liquid column that moves alternately and actuates a free wheel based mechanical translational to rotational displacement converter that propels the vehicle wheels. This configuration couples the translational movement of the Stirling engine pistons with the translational movement of the liquid column, avoiding normal forces that are associated with significant frictional losses at engine level.

The working principle of the Stirling-alternant hydraulic propulsion system is given in Fig. 1. The free piston Stirling engine 1 is driving the pistons of the hydraulic generator 3 that moves the liquid column in an alternant displacement to the freewheel motors 4 that propel the vehicle wheels 5. The hydraulic accumulator 2 has the role to adjust the energy delivered to the hydraulic pistons, to control the displacer movement of the Stirling engine and to store the energy recovered during the brake process. The directional control valve 6 assures a change in the succession process of the liquid column actuation, so that it is possible to change the rotational direction of the propelling wheel. The directional control valve 7 assures that the same flow is directed to the hydraulic motors for the case then road support is significant different for two or all four wheels.

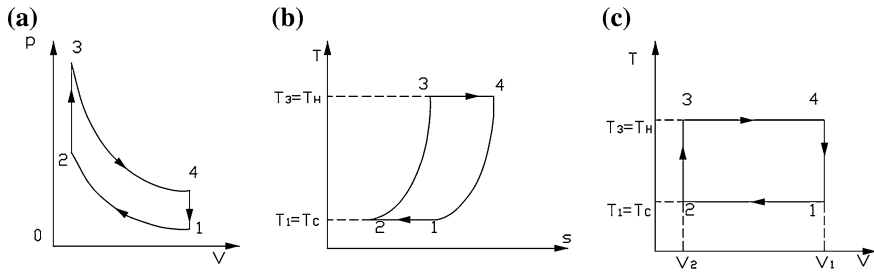


Fig. 2 Ideal working cycle of the Stirling engine: **a** pressure Vs. volume; **b** temperature Vs. entropy; **c** temperature Vs. volume

Otherwise, the parallel connection of the hydraulic consumers allows the speed that is installed due to road resistance, so that a safe cornering is assured.

2 Simulation of the Stirling Hydraulic Propulsion System

The objective of the simulation work was to have a set of possibilities to evaluate the potential of the Stirling engine and to couple the model with the hydraulic one. At first level, the thermodynamic model was analysed, considering the basic thermodynamic equations that compose the cycle, two isochoric and two isothermal functions, presented in Fig. 2.

In order to have a general overview of the thermodynamic processes and the elementary influences on system performance basic analytical models were used, having the parameters named in Table 1.

The equation of mechanical work is given by the equation:

$$L = mR \left[T_3 \ln \left(\frac{V_4}{V_3} \right) - T_1 \ln \left(\frac{V_1}{V_2} \right) \right] \quad (1)$$

The data identified for the working parameters in order to assure the system performance for vehicle propulsion are given in Table 2, having bolded the values considered for comparison.

It can be observed that a high potential for power density 163 kW/0.2 l working space at high efficiencies 55–60 % are theoretical possible. Considering the worst case efficiencies of fluid power propulsion of 80 %, overall propulsion efficiencies are in the range of 44 % significant above the actual performance of 25 % typical for vehicle propulsion systems.

The losses associated with the real model are cumulated and described by different authors that developed overall models for the Stirling engine. An overview on these models is given in (2). The equation confirmed by Beale [2], p. 99, also for free piston Stirling engines working in the temperature range 650–65 °C

Table 1 Nomenclature of parameters for thermodynamic analysis

Physical parameter	Symbol	Units	Physical parameter	Symbol	Units
Pressure	p	(N/m ²)	Volume of the regenerator	V _{reg}	(m ³)
Gas mass	M	(kg)	Temperature of the regenerator	T _{reg}	(°K)
Volume of hot zone	V _H	(m ³)	Power	P	(kW)
Temperature of hot zone	T _H	(°K)	Speed	N	(1/min)
Volume of cold zone	V _C	(m ³)	Mechanical work	L	(Nm)
Temperature of cold zone	T _C	(°K)	Max./min. working volume	V1/V2	(m ³)

Table 2 Influences of running parameters on the output power for the ideal Stirling thermodynamic power

m kg	R J/(kg·K)	t ₁ °C	t ₃ °C	V ₁ l	V ₂ l	L J	η [—]	N 1/min	P kW
0.05	287	100	10	0.1	0.2	895.1996	24	1000	15
0.1	287	100	10	0.1	0.2	1790.399	24	1000	30
0.1	287	100	10	0.1	0.2	1790.399	24	1000	30
0.1	287	500	10	0.1	0.2	9747.729	63	1000	162
0.1	287	350	10	0.1	0.15	3956.529	55	1000	66
0.1	287	350	10	0.1	0.2	6763.73	55	1000	113
0.1	287	350	10	0.1	0.2	6763.73	55	1000	113
0.1	287	350	10	0.1	0.2	6763.73	55	1500	169

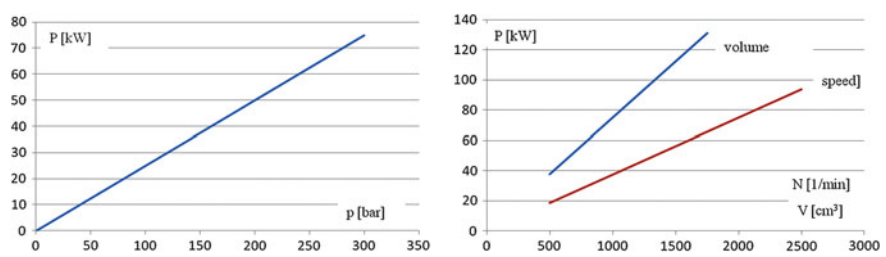


Fig. 3 Influence of running and design parameters on Stirling engine power output

shows the influence of different design and working parameters and results are plotted in figure N for the power range used for conventional vehicles.

$$P[kW] = 0.015 \times p[bar] \times \frac{N[\frac{1}{min}]}{60} \times V[cm^3] \tag{2}$$

The manner how running and design parameters are influencing the output power is given in Fig. 3.

It can be observed that the cylinder volume has a more important influence on power than speed. Power density can be obtained with high pressure and speed. Also acceptable running and design parameters are supporting system performance

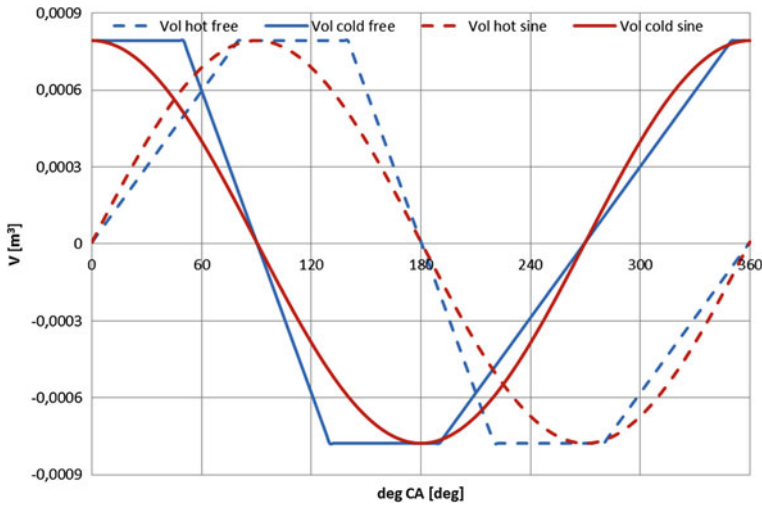


Fig. 4 Displacer and power piston movement for conventional sin-function and free shape design

using a real model equation as it is the proposal of Beale [2]. In this case supplementary measures, as proposed, like free shape of piston displacement and direct power transfer to the hydraulic liquid improve system efficiency. Due to limited compressibility of hydrogen, a parameter study may be performed if the higher heat capacity of hydrogen can be compensated by higher working pressures of another fluid.

2.1 Energy Balance Model

In order to identify the influence of piston speed on cycle energy balance, a zero-dimensional time dependent model was build up. The model is based on the energy balance of a control volume associated to the working space. The internal energy of the trapped gas si changed due to external interactions: change of mechanical work at displacer and power piston level and heat exchange during heating and cooling. The equations resulting from the energy balance and integrated into the perfect gas equation, has as result the mathematical time dependency of the pressure (Eq. 3), as functions of the time dependent interactions. Main time dependent influences are given by the displacement of the piston.

$$P = \frac{M.R}{\frac{V_H}{T_H} + \frac{V_C}{T_C} + \frac{V_{reg}}{T_{reg}}} \quad (3)$$

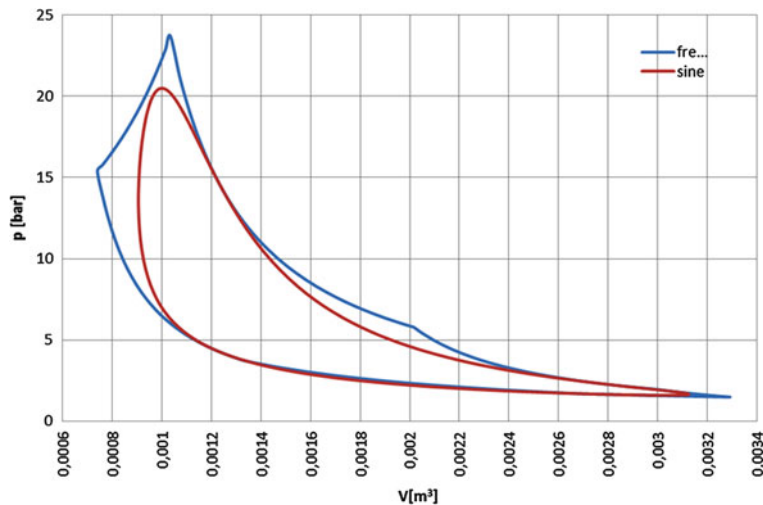


Fig. 5 Thermodynamic cycle of the Stirling engine for conventional and free shape piston displacement

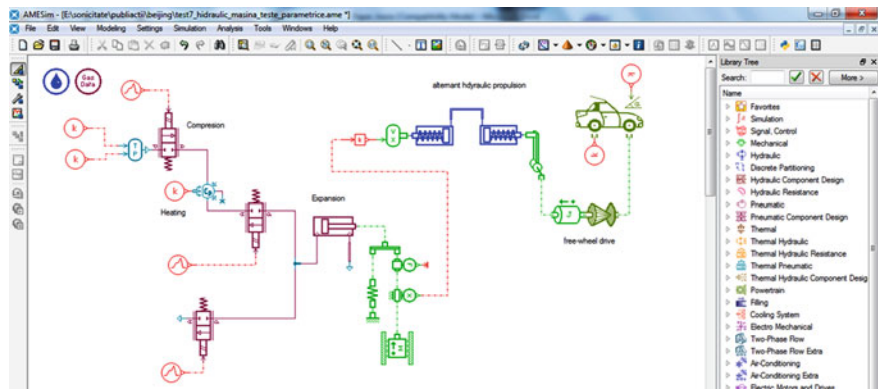


Fig. 6 Multi-domain model of the Stirling-alternant hydraulic propulsion system

Considering as reference, the sin-type displacement of the Stirling engine pistons having a 90° phase shift, a change of piston type of movement, as given in Fig. 4, has been applied. It can be observed, that only a small change of piston displacement, has a significant influence on the thermodynamic cycle, as can be observed in Fig. 5. The net mechanical work output in the case of conventional piston movement is 31.5 J and in the free displacement shape mode is 39.8 J, that mean an improvement of 21 %.

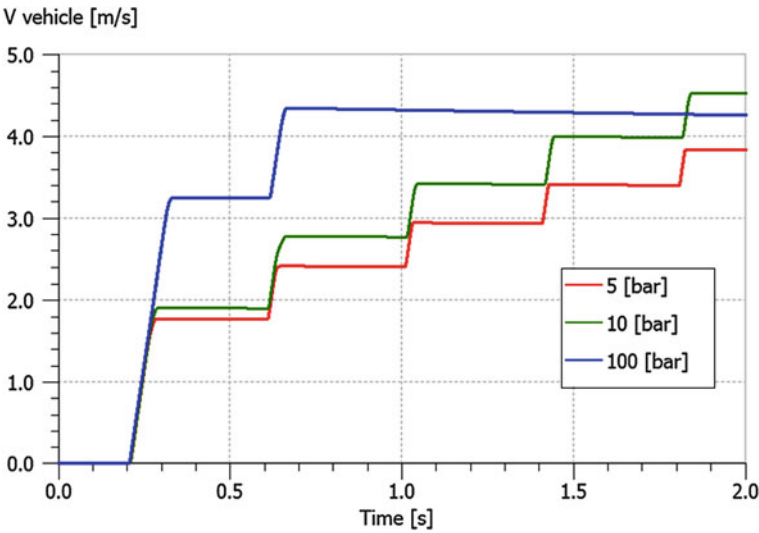


Fig. 7 Influence of Stirling engine working pressure on vehicle output speed-time dependency

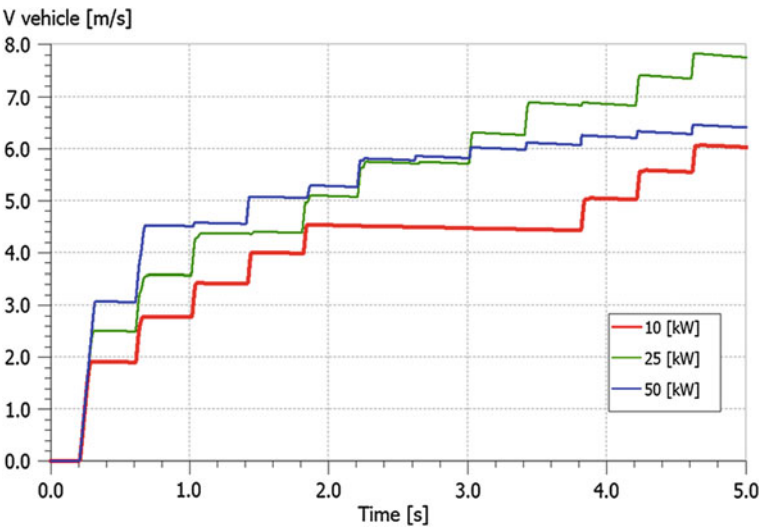


Fig. 8 Influence of added heat on vehicle performance

2.2 Multi-Domain Modelling and Simulation

For time-dependent analysis and coupling of the different subsystems from Stirling engine to the alternant hydraulic propulsion system coupled to a multi-domain analysis was performed using the model in Fig. 6.

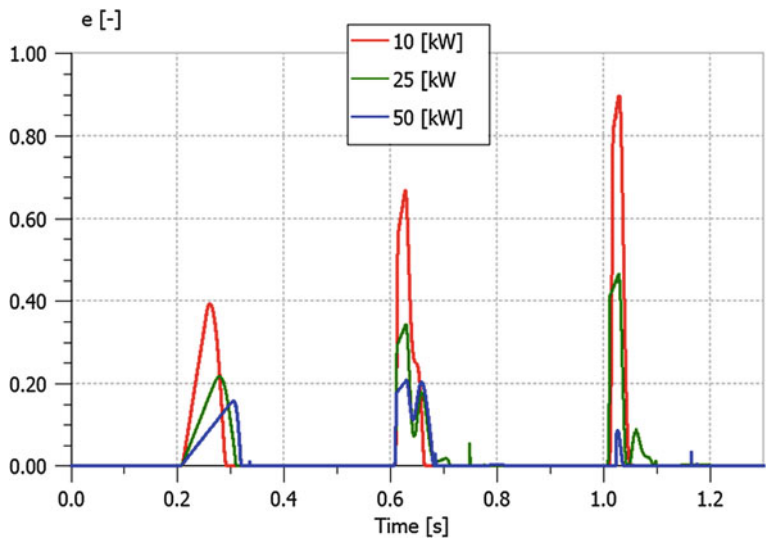


Fig. 9 Influence of time dependent power transfer on overall efficiency

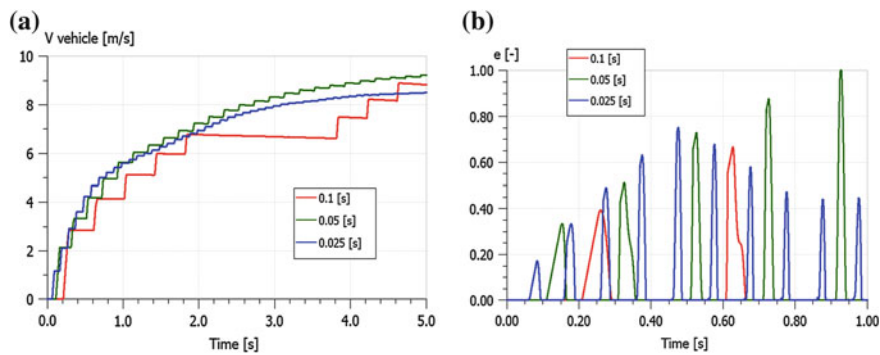


Fig. 10 Influence of actuation time on system performance: **a** vehicle speed; **b** efficiency

The model was build up considering the design structure indicated in [3, 4, 5] in order to allow a high flexibility of engine performance including directional control valves that can be actuated in a free manner, allowing the possibility of adjusting flow to the optimum heat transfer rate. The model of the Stirling-alternant hydraulic propulsion is based on the Stirling engine model composed of heat exchanger and pressure source with directional control valves, having the role of to assure the exposure of fluid to the heat exchange sources, the alternant hydraulic propulsion, based on two alternant moving cylinders maintained in a neutral position by mechanical springs, the translational-rotational movement device based on free wheel drive. The modelling parameters were adapted to an ATV type vehicle used also as demonstration vehicle, limited at 20 km/h (approximately 5 m/s).

Fig. 11 Temperature distribution in a heater applicable to the Stirling alternant hydraulic propulsion system

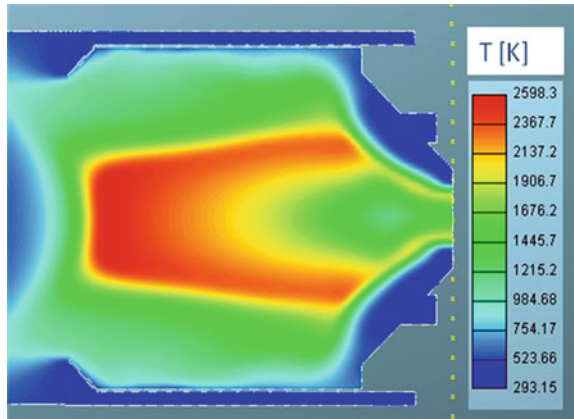


Fig. 12 The test rig and experimental vehicle to demonstrate the working principle of the Stirling alternant hydraulic propulsion system

Main influences of running parameters were analysed and studied bellow. Considering the influence of the working pressure of the Stirling engine on vehicle speed as objective parameter, Fig. 7, it can be observed that a higher pressure assures a rapid acceleration, but top speed remains limited. This capability is given by the induced pressure at working piston level—25 to 9 bar.

The added heating power has also a significant influence on system performance. Higher heat flows assure a very rapid start-up, but top speed is reached with a moderate amount of energy added to the system.

The analysis of the overall efficiency computed by dividing the output wheel power to the heat added to the working gas is plotted in Fig. 9. It can be observed that a medium amount of heat is transferred with increasing efficiency. It can be observed from the data in Figs. 8 and 9, that making available a great amount of energy is not necessary to be absorbed by the system. Also the data show that an impulsive energy transfer is very effective.

Table 3 Comparison of computed and measured data for the Stirling alternant hydraulic propulsion system

	Gas pressure (bar)	Actuation pressure (bar)	Hydraulic pressure (bar)	Working Frequency (Hz)	Vehicle speed (m/s)
Computed	6	7	14.0	10	8
Measured	7	8	13.5	10	7
Error (%)	14	12.5	3.7	Adopted	14.28

A similar analysis is synthesised in Fig. 10, considering the actuation time as parameter. It can be observed that there it is an optimum time to expose the working fluid to heat transfer. A high energy transfer rate assures a smoother displacement of the vehicle.

In order to identify the possibility of achieve the necessary temperatures, a CFD 3D simulation was performed for a conventional heater based on natural gas as fuel, considering the simulation principles after [6]. The temperature distribution is presented in Fig. 11.

3 The Experimental Model

The experimental model of the Stirling alternant hydraulic propulsion system was developed at test rig level and vehicle level. The images of the test rig and of the vehicle are given in Fig. 12, and a comparison of computed and measured data is given in Table 3.

4 Conclusions

The analysis of the Stirling alternant hydraulic principle has been proven to be a system able to run, creating a set of possibilities to apply optimisation measures for both, the Stirling and the fluid power transmission principle. The free piston solution has the potential of a 21 % efficiency increase due to the free shape of the piston displacement curve. An additional 5 % improvement is expected due to the absence of the normal forces that induce piston friction. It has been proven that impulsive energy transfer is very efficient. The overall layout allows compensating current drawbacks of the Stirling engine like start/stop behaviour and control, due to the hydraulic accumulator.

References

1. Pischinger S (2009) Alternative vehicle propulsion systems. Lecture Notes, Aachen
2. Martini WR (2004) Stirling engine design manual. University Press of the Pacific, Honolulu
3. Woods RL, Kent L (1997) Modeling and simulation of dynamic systems. Prentice Hall, New Jersey, Upper Saddle River
4. Abaitancei H (2010) Sisteme de recuperare hidraulica a energiei la bordul autovehiculelor. University Transilvania Publishing House, Brasov
5. LMS Amesim Tutorial Manual. Release 10
6. AVL Fire Tutorial Manual